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Analysis of an Innovative Microchannel Condenser Design for Modular Chillers and Unitary Rooftop Air Conditioners

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ABSTRACT

An innovative way of integrating microchannel condensers in modular chillers and rooftop air conditioners is presented in this paper. The geometrical characteristics of the heat exchanger and how it can be incorporated into commercial air conditioners is discussed. Details are given on the performance analysis of the heat exchanger under typical chiller operating conditions. A simulation study is presented comparing the proposed coil and the conventional microchannel coil. Analysis shows that the new heat exchanger increases the capacity of a typical chiller condenser by 20% at same foot print. When operated using the same fan as baseline, for the same capacity the new heat exchanger resulted in a 2 ^{o}C drop in condensing temperature. Testing was performed at a small scale coils to validate the gains predicted by the simulation. The testing showed that the new heat exchanger can deliver a 26% higher heat rejection rate. At same heat rejection rate the testing showed that the new heat exchanger operated at 1.9 ^{o}C lower condensing temperature.

1. INTRODUCTION

Buildings, both commercial and residential, account for nearly 40% of the total energy consumption in the US. The share of commercial buildings increased from 14% in 1980 to around 18% in the year 2005 (US Department of Energy, 2008). As the population grows and moves more towards urban areas, this figure is expected to only grow in the coming years. Air conditioning, which is a major consumer of energy in buildings is also a significant contributor to global warming, either directly through the use and subsequent leakage of refrigerants or indirectly via power consumption. Direct contributions to global warming have been reduced by environmental regulations, such as F-gas regulations or global treaty agreements, such as the Kyoto protocol and Montreal protocols (European Union, n.d.). The indirect contributors are mitigated by national and regional energy efficiency regulations.

Recently the chiller efficiency requirements were increased to comply with Addendum Ch to ASHRAE 90.1 (ASHRAE, 2010), as shown in Table 1. Similarly, the DOE is proposing an increase in minimum efficiency requirements of large rooftops, as shown in Table 2. In both the chiller and rooftop cases, meeting the targets would require significant technological advancements.

A new method to incorporate the microchanel heat exchangers in modular chillers and commercial rooftops is presented here. A simulation study will be presented to demonstrate the benefits of the proposed heat exchanger design over conventional designs. Results will be presented from the testing of the conventional microchannel heat exchanger and the new design.

Table 1: Increase in chiller efficiency requirements due to Addendum Ch, ASHRAE 90.1

Size		Current	Path A	Path B
< 150*TR*	EER	9.6	10.1	9.7
	IPLV	12.5	13.7	15.8
> 150*TR*	EER	9.6	10.1	9.7
	IPLV	12.8	14.0	16.1

Table 2: Proposed mandatory efficiency requirements (IEER) for rooftop A/C

Size	Current[1]	2018	2023
$\geq 65,000 Btu/h$ & $< 135,000 Btu/h$	11.2	12.9	14.8
$\geq 135,000 Btu/h$ & $< 240,000 Btu/h$	11.0	12.4	14.2
$\geq 240,000 Btu/h$ & $< 760,000 Btu/h$	10.0	11.6	13.2

[1] Current regulations do not include IEER but only uses an EER

2. EFFECT OF CONDENSERS ON SYSTEM PERFORMANCE

For systems with air-cooled condensers, the condenser performance impacts the overall system performance in multiple ways. Firstly, an increase in heat exchanger performance results in a reduced condensing temperature, which in turn reduces compressor power. Secondly, the allowable operating envelope of the compressors gets widened to a higher ambient temperature. Finally, the superior condenser performance lowers the liquid temperature which increases both the capacity and efficiency of the system.

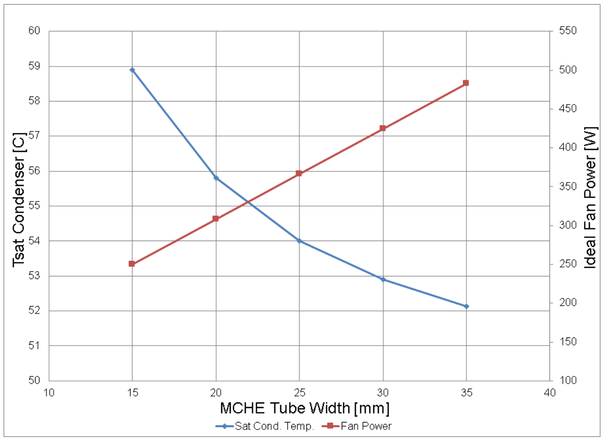
The power of the compressor, and consequently the efficiency of the system as a whole, is dictated by the condensing temperature. The heat transfer performance of a condenser is dictated by Equation (1). As seen in the equation, for a given capacity, the condensing temperature can be decreased by increasing the *UA* value. Recently, microchannel heat exchangers (MCHE) have become popular in chiller and rooftop systems. Compared to the conventional fin-and-tube heat exchangers, microchannel heat exchangers offer better thermo-hydraulic characteristics, thereby increasing the overall heat transfer coefficient (U). However, a drastic increase in the overall heat transfer coefficient to meet the stricter requirements is typically expensive and takes a long time. Thus the most straightforward way to increase heat exchanger performance is to increase the heat transfer surface area.

$$\dot{Q} = UA\Delta T_m \tag{1}$$

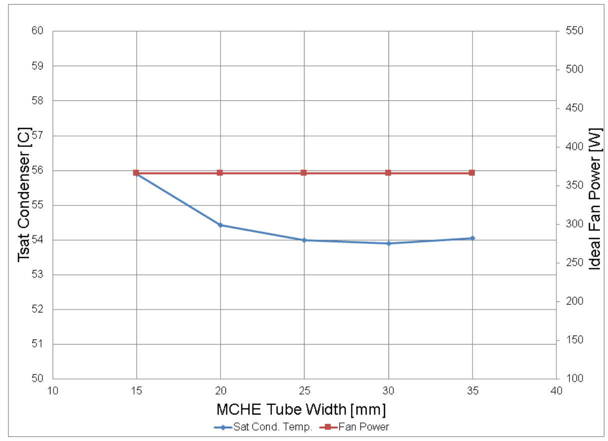
2.1 Increasing Depth of HX

As noted in the previous section, increasing the heat transfer area is an easy way to achieve improve efficiency. One way to increase the heat transfer area is to increase the depth of the heat exchanger. In conventional fin-an-tube heat exchangers, this is achieved by adding tube rows in the direction of airflow. In microchannel heat exchangers, a wider tube is used to increase the depth of the heat exchanger. Increasing the depth of heat exchanger increases the heat transfer area, which has a positive effect on the performance of the heat exchanger. On the other hand, a deeper heat exchanger results in a higher pressure drop on the air side. Unless the condenser fan is designed for this higher pressure drop, the resulting airflow through the coil will be lower, resulting in a lower air side heat transfer coefficient. Depending on the system balance, a drop in the air side heat transfer coefficient will negate any gain in the surface area. If the designer chooses to redesign the fan to provide the same air flow as the original coil, then the power consumption of the condenser fan will invariably increase. The result is that in some cases there will be no system efficiency gain since any lower compressor power due to reduced condensing temperatures will be compensated by higher condenser fan power.

The information presented in this section is obtained from a simulation study using CoilDesigner software (Jiang, Aute, & Radermacher, 2006). The simulation analysis presented here was done using refrigerant R-410A. The saturated condensing temperature was varied to attain the same capacity with different coil depths. The inlet superheat of the refrigerant was 17 ^oC and outlet liquid temperature was 40.5 ^oC. The conditions of the air entering the heat exchanger was taken as 35 ^oC DB at sea level and 50% relative humidity. Details of the microchannel coil used for this study is given in Yanik and Padhmanabhan (2015). Figure 1(a) and Figure 1(b) show the effect of the depth of the heat exchanger on the condensing temperature at the same airflow and the same fan power, respectively. The condenser fan power increased as the depth increased for the same airflow, as can be seen in Figure 1(a), where as the drop in condensing temperature starts to drop as depth increases. If the fan power is kept constant, then the condensing temperature will not drop significantly as the depth increases. As a result, it can be concluded that the strategy of increasing depth will result in diminishing returns in most cases and in some cases will even result in a drop in overall efficiency.

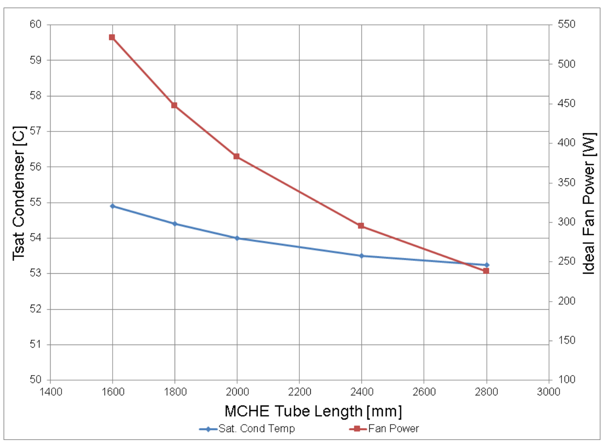


(a) At same airflow

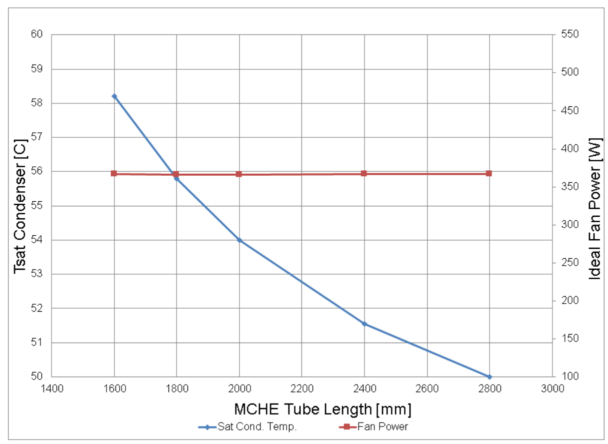


(b) At same fan power

Figure 1: Effect of increase in depth on condensing temperature



(a) At same airflow



(b) At same fan power

Figure 2: Effect of increase in face area on condensing temperature

2.2 Increasing Face Area

Another way to increase the heat transfer surface area is by increasing the coil face area either by increasing the coil height or coil/tube length. If the designer chooses to keep the condenser fan same, then the resultant airflow with a larger face area heat exchanger will be higher. Depending on the fan characteristics, the increase in the air face velocity could be higher or lower, causing the airside heat transfer coefficient to increase or decrease.

A simulation analysis similar to the previous section was done with varying the tube length of the heat exchanger. Figure 2(a) and Figure 2(b) show the effect of the tube length (face area) of the heat exchanger on the condensing temperature at the same airflow and same fan power, respectively. As explained in the previous paragraph, as face area increases with the same airflow, the heat transfer coefficient drops. As a result, the condensing temperature will not drop significantly since any gain by an increase in heat transfer area is compensated by a drop in overall heat transfer coefficient. This is evident in Figure 2(a). In contrast to the increasing depth option, the increase in face area will result in lower power draw at the same airflow. As explained by Webb and Kim (2005), the right way to compare two heat exchangers/fan systems is to keep the fan power constant. Figure 2(b) shows the variation in condensing temperature with tube length when the fan power is kept constant. By keeping the fan power constant, a greater reduction in condensing temperature is achieved. This is due to the fact that with a higher face area when the fan power is kept constant, the fan/heat exchanger balance point would be at a higher air flow rate resulting in a higher heat transfer coefficient.

3. NEW MCHE DESIGN FOR MODULAR A/C

Although it is clear from the previous section that increasing the coil face area is very attractive in terms of increasing coil performance and system efficiency, it is not always practical from a commercial point of view. Traditionally, increasing the coil face area meant increasing the size of the unit which increased the cost of manufacturing, shipping, and warehousing. Moreover, this kind of approach is not suitable for the replacement market where a unit's size and weight are critical.

Many modern commercial roof top air conditioning systems and air cooled chillers utilize a modular condenser with V-sections as shown in Figure 3. In this design, the condenser coils are separated by a V-shaped panel that prevent air from bypassing the coils. This V-panel constitutes a large portion of the face area and is not utilized for heat transfer purposes. Moreover, the air has to make sharp turns and has to pass through small openings between V-panels to flow across the heat exchangers in middle. This undesirable flow path results in the fan operating at a non-optimal point. Conventional fin-and-tube designs do not allow the effective use of the V-panel area due to manufacturing constraints. MCHE heat exchanger with brazed aluminum fins and tubes allow the coil to be formed into shapes that are not practical with traditional fin and tube coils. One of these configurations is an MCHE coil that utilizes the V-panel area as a heat transfer surface as shown in Figures 4(a) and 4(b). We will refer such a coil as MCHE V-coil.

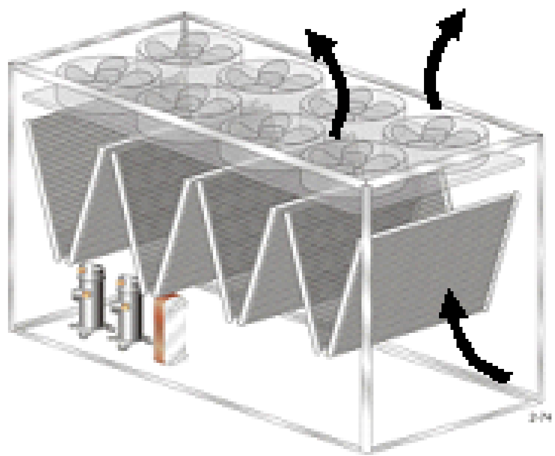
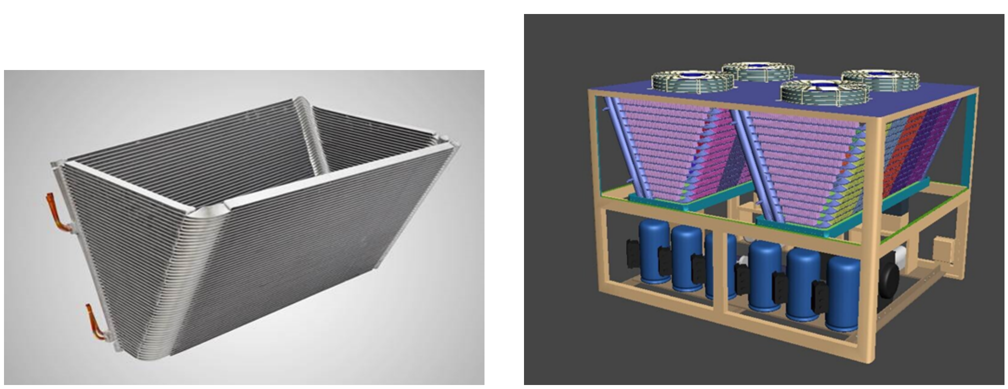


Figure 3: Schematic of a modular chiller

The manufacturing process of the coil is not trivial since the design depends a lot on the V-panel dimensions. Figure 5 shows a schematic of the V-coil with associated dimensions and angles. The V-panel is defined by the angle between



(a) Individual V-Coil module   (b) Schematic of a chiller with V-Coil

Figure 4: V-coil Module and Assembly

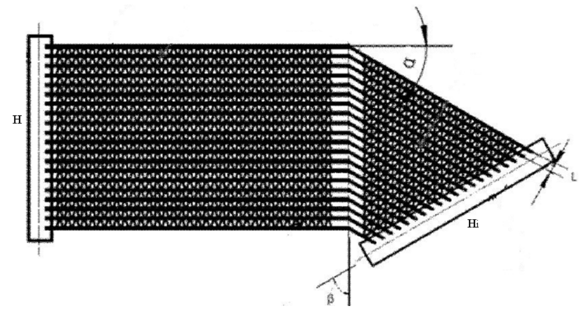


Figure 5: Geometry of the Proposed Microchannel Heat Exchanger

the two inclined manifolds and is denoted by the V-angle (β). The coil is manufactured by folding the V-panel area by 90^o with respect to the straight portion of the coil. Manufacturing of this coil, however, requires that the tubes in the V-area be bent with reference to the straight tubes before folding. The amount of bend for the tube portion in the V-panel is defined as angle α. It can be shown by geometric analysis that $\beta = 2\alpha$. The V-coil design allows designers to increase the heat transfer area of the heat exchanger without increasing the unit foot print. The designer can utilize the increase in area in different ways depending on the objectives. For example:

- Increasing the capacity of the unit without increasing the size.
- Increasing the efficiency of a unit without increasing the footprint.

4. RESULTS AND DISCUSSION

4.1 Simulation

The microchannel coil in a commercially available chiller was modeled using CoilDesigner for the analysis presented in this section. A V-coil designed to fit in the same frame is compared with the baseline slab microchannel coil. In order to analyze the performance of the V-coil, an equivalent slab coil with the same face area as the V-coil was used. The height of the equivalent coil was kept same as the height of the V-coil. The length of V-coil was then calculated as shown in Equation (2).

$$L_{eq} = \frac{A_{vcoil}}{H_{vcoil}} \tag{2}$$

The ambient condition used for all analyses is 35 ^{o}C, 50% RH. As the V-coil has varying tube lengths across its height, air flow across the coil varies. A similar situation occurs in the refrigerant side also for each tube. The analysis thus accounts for non-uniform airflow and refrigerant distribution. This is achieved by dividing the coil into three parts in the direction of manifold height. Each part is analyzed using an equivalent slab coil approach. The resultant air flow and refrigerant flow through each section is determined by equalizing the pressure drop, as shown in Equation (3)

$$\Delta P_{a,1} = \Delta P_{a,2} = \Delta P_{a,3} \qquad \Delta P_{r,1} = \Delta P_{r,2} \tag{3}$$

Figure 6 shows the result of the analysis for the slab coil and V-coil. The analysis was done for two different pitch angles for the condenser fans. The baseline case for the analysis is the slab coil operating with a fan pitch of 27^o. As can be seen in the figure, the V-coil operating with the same fan as the baseline resulted in a 10% higher air flow. It is interesting to note that the fan power consumption for the V-coil is lower than the baseline even though the airflow increases. This is very advantageous from a system design point since the designer can use a fan that draws the same power as the baseline to increase the airflow further, resulting in an even increased performance. This is demonstrated in the figure for the V-coil using a fan pitch of 30^o. In the case of the V-coil operating at fan pitch of 30^o, the resulting airflow is 17% higher than the base case.

The V-coil allows the designer flexibility in how it can be incorporated in the system design. Table 3 shows the results of slab coil and V-coil under various objectives. The baseline slab coil at a fan pitch of 27^o has a heat rejection rate of 101 kW at a condensing temperature of 51.7 ^{o}C. A V-coil running at the same fan pitch and condensing temperature results in a 12% increase in heat rejection rate with a 7% drop in fan power. For the same capacity as the baseline, the

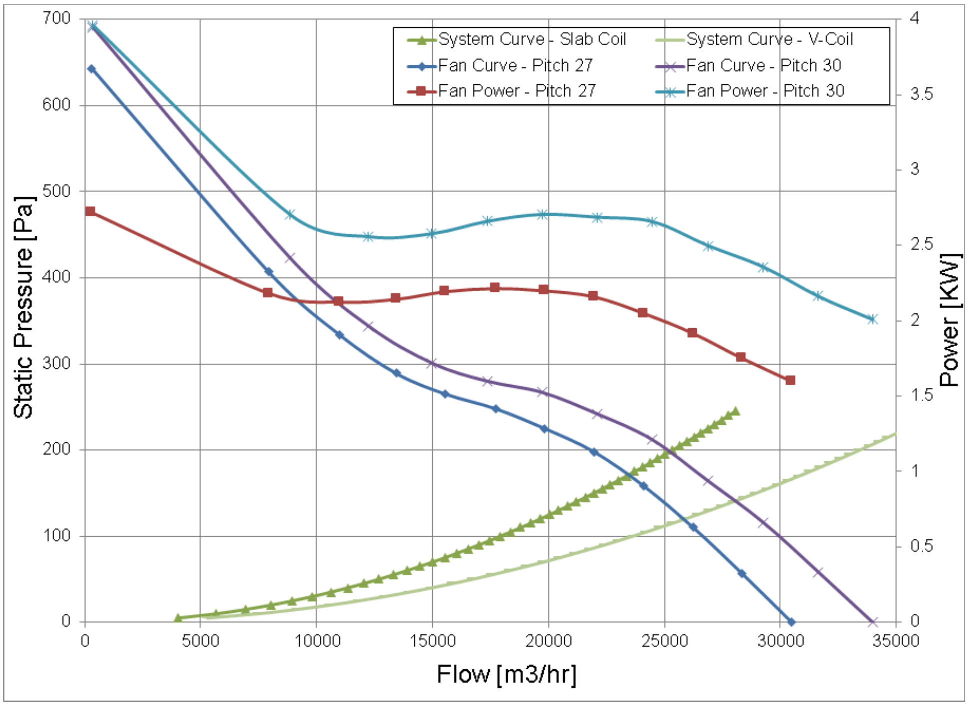


Figure 6: Comparison of a slab coil and V-coil at two fan pitch

Table 3: Slab coil Vs V-coil at various fan pitches

Coil Type	**Fan Pitch** [^{o}]	**Air Flow** [m^3/hr]	**Fan Power** [KW]	**Cond. Temp** [^{o}C]	**Capacity** [KW]	**Spec. Power** [KW/ton]
Slab	27	23,452	2.1	51.7	101.3	0.021
V-Coil	27	25,807	1.95	51.7	113.2	0.017
V-Coil	27	25,807	1.95	50.0	103.0	0.019
V-Coil	30	27,494	2.6	51.7	121.5	0.021
V-Coil	30	27,494	2.6	48.8	101.2	0.026

V-coil can run at a 1.7 ^{o}C lower condensing temperature, which improves efficiency. As explained before, the drop in fan power can be utilized to improve the performance further. By using a fan pitch of 30^o, the heat rejection can be increased by 20% at the same condensing temperature as that of the baseline. For the same heat rejection rate as the baseline, the V-coil can run at a 3 ^{o}C drop in condensing temperature. It should be noted that the specific power consumption (kW/ton) drops for the V-coil when the fan pitch is the same as the baseline. Even though the fan power consumption goes up at a higher fan pitch, the specific power consumption is the same as the baseline indicating that the increase in fan power yields a proportional increase in capacity.

Figure 7 shows the UA of the slab coil and V-coil along with the fan power. The advantages of using a V-coil when using the same fan can be seen clearly in the figure. For example, for a baseline power draw of 4.0 kW, the baseline slab coil results in a UA value of around 8.1 KW/^{o}C, while a V-coil at same fan power results in around 9.9 kW/^{o}C. This increase in UA translates into a proportional reduction in condensing temperature, thereby reducing the compressor power consumption.

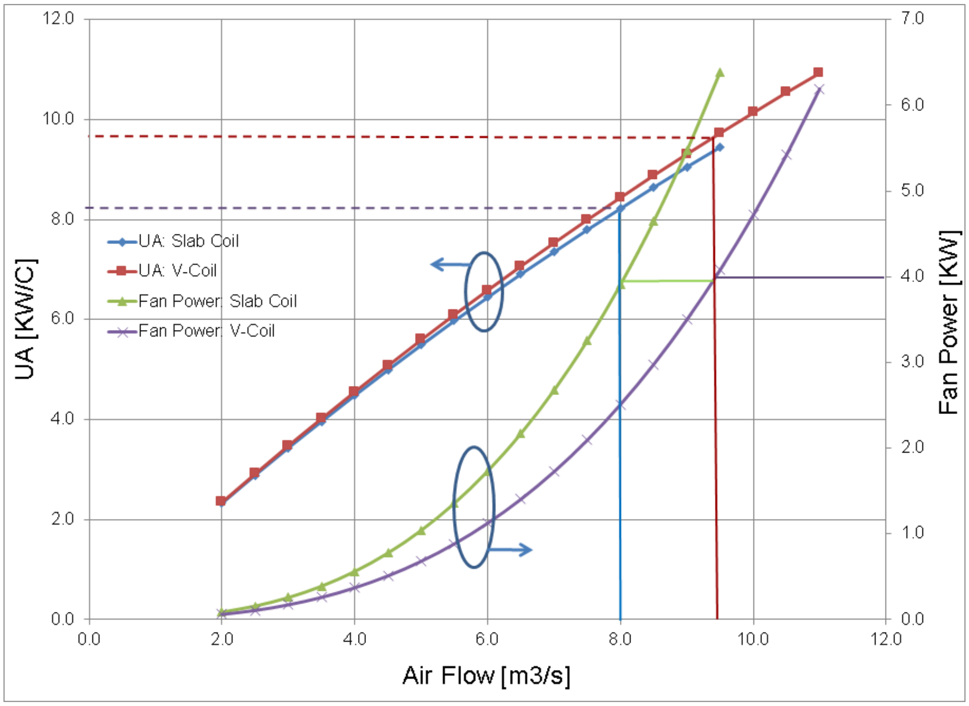


Figure 7: Comparison of a slab coil and equivalent V-coil

4.2 Testing

Testing was carried out to evaluate the performance gains achievable by V-coils. Due to testing constraints the coils used were smaller than the ones used in the simulation study presented in the previous section. In order to make the comparison meaningful the results from the testing are non-dimensionalized, so that the gain seen in testing can be compared with gain predicted by simulation.

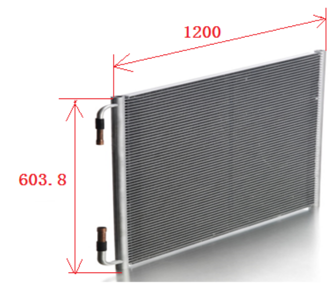
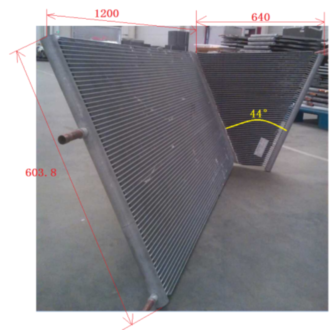
The details of the slab coil and the V-coil used for testing are shown in Table 4. The slab coil and V-coil were tested individually at similar conditions in a psychometric wind tunnel. Results from the test are shown in Figure 9. The figure shows that when run at the same airflow, the V-coil provides around 7% higher heat rejection rate. When run at the same pressure drop as the slab coil, then the V-coil delivers a 26% higher heat rejection capacity.

Table 5 shows the test results. It can be seen that for the same heat rejection rate as the slab coil, the V-coil ran at around a 1.9 ^{o}C lower condensing temperature. This lower condensing temperature will result in a lower compressor power, thereby resulting in a more efficient system. The test results match the predictions of the analysis to a very high degree.

Slab Coil "V" Coil

Figure 8: Test setup to compare slab coil and V-coil

Table 4: Geometry Information of slab coil and V-coil

	Slab coil	**V-coil**
Configuration	2 Pass (41/18)	2 Pass (41/18)
Tube Length (Top) [mm]	1200	1840
Manifold Height [mm]	603.8	603.8
Image		

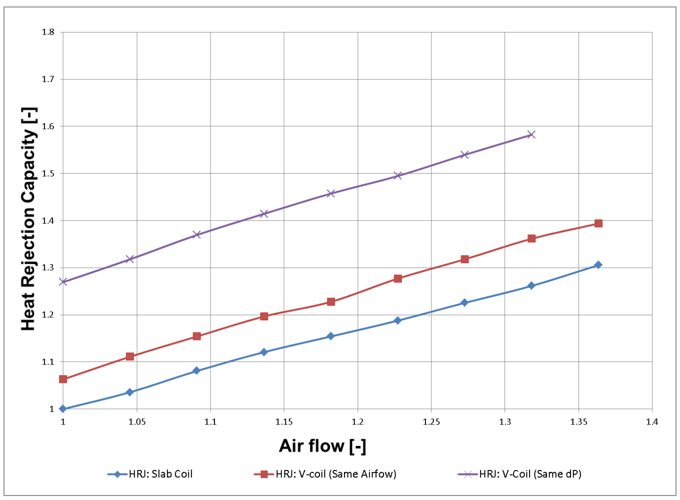


Figure 9: Tested performance of slab Coil & V-Coil

5. CONCLUSIONS

A novel method of using microchannel heat exchangers in commercial rooftop units and modular chillers was presented in this paper. The V-coil technology allows the system designers to achieve the increased efficiency requirements without increasing the foot print. The V-coil when run with the same condensing temperature as the slab coil delivered a 12% higher heat rejection rate with a 7% drop in fan power. When the specific fan power (kW/ton) is kept constant then the V-coil delivered a 20% higher heat rejection rate. The performance of the V-coil was further improved by increasing the fan pitch to provide more airflow. The testing performed to compare the performance of the V-coil and the slab coil showed similar gains as predicted by the simulation. From the analysis it can be concluded that the use of V-coil is an effective strategy to meet the increased efficiency requirements for commercial rooftops and modular chillers.

Table 5: Tested Performance of Slab coil Vs V-coil †

Coil Type	Air flow [-]	Cond. Temperature [^oC]	Subcooling [K]	Capacity [-]
Slab	1	50.0	9	1
V-Coil	1.1	50.0	9	1.16
V-Coil	1.1	48.1	9	1

† Airflow & Capacity values are non-dimensionalized based on the slab coil

NOMENCLATURE

A	Heat Transfer Area	(m^2)
EER	Energy Efficiency Ratio	$(\frac{BTU/hr}{W})$
H	Height	(m)
IPLV	Integrated Part Load Value	$(\frac{BTU/hr}{W})$
ΔP	Pressure drop	(Pa)
$\dot{Q}$	Heat Transfer Rate	(kW)
ΔT	Temperature Difference	(K)
U	Overall Heat Transfer Coefficient	$(kW/m^2 - K)$

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